Chapter 5
Efficiencies of Power Plants

5.1 Introduction

To assist in improving the efficiencies of power plants, their thermodynamic characteristics and performance are usually investigated. Power plants are normally examined using energy analysis but, as pointed out previously, a better understanding is attained when a more complete thermodynamic view is taken, which uses the second law of thermodynamics in conjunction with energy analysis via exergy methods.

Although exergy analysis can be generally applied to energy and other systems, it appears to be a more powerful tool than energy analysis for power cycles because it helps determine the true magnitudes of losses and their causes and locations, and improve the overall system and its components. In this chapter, we provide an overview of various energy- and exergy-based efficiencies used in the analysis of power cycles, including vapor and gas power, cogeneration and geothermal power plants. Differences in design aspects are considered. The various approaches that can be used in defining efficiencies are identified and their implications discussed. Numerical examples are provided to illustrate the use of the different efficiencies, and the results include combined energy and exergy diagrams.

Note that the emphasis in this chapter is to describe various energy- and exergy-based efficiencies used in power plants and discuss the implications associated with each definition. Therefore, simple cycles are selected to keep the complexity of the plants at a minimum level for gas and vapor cycles the better to facilitate understanding of the efficiencies, which can be very useful for improved energy management in power plants. One can easily adapt the efficiencies discussed here to more complex power systems. Some efficiency definitions for gas cycles found in many thermodynamics textbooks are repeated so that the coverage in this chapter is comprehensive and can serve as a convenient and practical tool for students, engineers, and researchers. It is shown that a better understanding of energy and exergy efficiencies and their successful use can help improve energy management in power plants [6].

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5.2 Efficiencies of Vapor Power Cycles

The thermal efficiency, also referred to as the energy efficiency or the first law efficiency, of a power cycle is defined as

$$\eta_{\text{th}-1} = \frac{w_{\text{net, out}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} \tag{5.1}$$

where $w_{\text{net,out}}$ is the specific net work output, q_{out} is the specific heat rejected from the cycle, and q_{in} is the specific heat input to the cycle, which is usually taken to be the specific heat input to the steam in the boiler of a steam power plant. That is,

$$q_{\text{in}} = h_3 - h_2 \tag{5.2}$$

where h denotes specific enthalpy and the subscripts refer to state points in Fig. 5.1. This simple approach neglects the losses occurring in the furnace–boiler system due to the energy lost with hot exhaust gases, incomplete combustion, and so on. To incorporate these losses, one can express the thermal efficiency of the cycle by a second approach as

$$\eta_{\text{th}-2} = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{fuel}} q_{\text{HV}}} \tag{5.3}$$

where $\dot{m}_{\text{fuel}}$ is the mass flow rate of fuel and q_{HV} is the heating value of the fuel, which can be chosen as the higher or lower heating value. For furnace–boiler systems where the water in the exhaust gases is not expected to condense, as in internal combustion engines, it is customary to use the lower heating value [22].

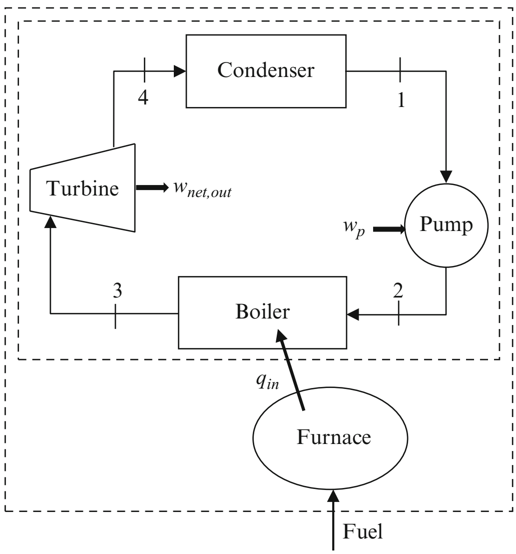

Fig. 5.1 Simple steam power plant

Some tend to use lower heating values to make a device appear more efficient. This is frequently done in manufacturer descriptions of commercial boilers. Often a claimed efficiency exceeds 100%, as discussed in Chap. 4. This is due to recovering some of the heat of condensation of steam in the exhaust gases while still defining boiler efficiency based on the lower heating value. This is misleading and a thermodynamically improper use of efficiency. If there is any possibility of recovering some of the energy of condensing steam in exhaust gases, the efficiency should be based on the higher heating value.

The second law efficiency, also referred to as exergy efficiency, of a power-producing cycle is defined as

$$\varepsilon = \frac{w_{\text{net, out}}}{x_{\text{in}}} = 1 - \frac{x_{\text{dest}}}{x_{\text{in}}} \tag{5.4}$$

where x_{in} is the specific exergy input to the cycle and x_{dest} is the specific total exergy destruction in the cycle. One can express the exergy input to the cycle as the exergy increase of the working fluid in the boiler of a steam power plant (Fig. 5.1) as

$$x_{\text{in}} = h_3 - h_2 - T_0(s_3 - s_2) \tag{5.5}$$

where T_0 is the dead-state or environment temperature and s is the specific entropy. Substituting (5.5) into (5.4),

$$\varepsilon_1 = \frac{w_{\text{net, out}}}{h_3 - h_2 - T_0(s_3 - s_2)} \tag{5.6}$$

In this definition, the irreversibilities during energy transfer from the furnace to the steam in the boiler are not accounted for. Alternatively, the exergy input to the cycle may be defined as the exergy input to the boiler accompanying the heat transfer. The exergy efficiency in this case becomes

$$\varepsilon_2 = \frac{w_{\text{net, out}}}{q_{\text{in}}\left(1 - \dfrac{T_0}{T_s}\right)} \tag{5.7}$$

where T_s is the source temperature, which is the temperature of the heat source (i.e., furnace), and q_{in} is given by (5.2). This efficiency definition incorporates the irreversibility during heat transfer to the steam in the boiler. We may also incorporate in the efficiency definition the exergy destruction associated with fuel combustion and the exergy lost with exhaust gases from the furnace. In this third approach, the exergy efficiency can be expressed as

$$\varepsilon_3 = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{fuel}} x_{\text{fuel}}} \tag{5.8}$$

where x_{fuel} is the specific exergy of the fuel. The exergy of a fuel may be obtained by writing the complete combustion reaction of the fuel and calculating the reversible work by assuming all products are at the state of the surroundings. Then the exergy of fuel is equivalent to the calculated reversible work. For fuels whose combustion reaction involves water in the products, the exergy of the fuel is different depending on the phase of the water (vapor or liquid). The exergies of various fuels listed in [10] are based on the vapor phase of water in combustion gases.

Different efficiency definitions are possible if one selects different system boundaries. Clearly defining the system boundary allows the efficiency to be defined unambiguously. For example, the exergy efficiencies in (5.7) and (5.8) correspond to systems whose boundaries are given by the inner and outer dashed lines, respectively, in Fig. 5.1.

Example 5.1 A numerical example is used to illustrate and contrast the various efficiencies defined in this section. We consider a simple steam power plant with a net power output of 10 MW and boiler and condenser pressures of 10,000 and 10 kPa, respectively (Fig. 5.1). We assume a turbine inlet temperature of 500°C and isentropic efficiencies of 85% for both the turbine and the pump. In addition, we assume that the furnace–boiler system has an efficiency of 75%. That is, 75% of the lower heating value of the fuel is transferred to the steam flowing through the boiler and the remaining 25% is lost, mostly with the hot exhaust gases passing through the chimney. The source and sink temperatures in (5.7) are taken as 1,300 and 298 K, respectively. We consider methane as the fuel with a lower heating value of 50,050 kJ/kg and a chemical exergy of 51,840 kJ/kg [10].

For the given values and assumptions, an analysis of this cycle yields

$$w_{\text{net, out}} = 1{,}081\,\text{kJ/kg},\; q_{\text{in}} = 3{,}172\,\text{kJ/kg},\; x_{\text{in}-1} = 1{,}400\,\text{kJ/kg},\; x_{\text{in}-2} = 2{,}444\,\text{kJ/kg}$$

as well as the following efficiency values:

$$\eta_{\text{th}-1} = 34.1\%\;,\; \eta_{\text{th}-2} = 25.6\%,\; \varepsilon_1 = 77.2\%,\; \varepsilon_2 = 44.2\%,\; \varepsilon_3 = 24.7\%$$

When the energy and exergy losses in the furnace–boiler system are not considered, the thermal efficiency is 34.1% whereas the corresponding exergy efficiency is much higher (77.2%). However, when the losses in the furnace–boiler are considered, the exergy efficiency (24.7%) is lower than the thermal efficiency (25.6%). When teaching undergraduate thermodynamics, it is normally stated that the exergy efficiency is greater than the thermal efficiency for heat engines, referring to the first approach here. This point is made by emphasizing that thermal efficiency is the fraction of heat input that is converted to work whereas exergy efficiency is the fraction of the work potential of the heat (this work potential, i.e., exergy, is smaller than heat) that is converted to work. However, when one considers the effect of furnace–boiler losses, and uses the chemical exergy of the fuel in the exergy efficiency and the heating value of the fuel in the thermal efficiency, the exergy efficiency becomes smaller than the

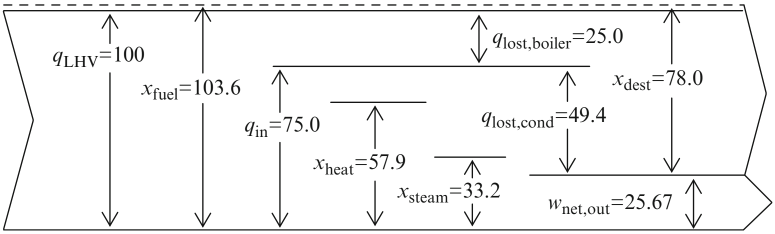

Fig. 5.2 Combined energy and exergy diagram for the steam power plant considered

thermal efficiency. In thermodynamics, it is often misleading to make generalized statements as they may not always apply. For example, can we state that the exergy efficiency, based on the third approach in (5.8) (ε_3), is always lower than the thermal efficiency as defined by the second approach in (5.3) ($\eta_{\text{th-2}}$)? The answer will be yes only if the chemical exergy of the fuel is always greater than its heating value. According to data in [10], this is the case for methane but not for hydrogen (q_{LHV} = 119,950 kJ/kg, x_{fuel} = 117,120 kJ/kg).

For a reversible heat engine cycle operating between a source at T_s and a sink at T_0, the thermal efficiency is given by

$$\eta_{\text{th, rev}} = 1 - \frac{T_0}{T_s} \tag{5.9}$$

The ratio of the actual thermal efficiency to the thermal efficiency of a reversible heat engine operating between the same temperature limits gives a type of exergy efficiency of the heat engine. For a furnace temperature of T_s = 1,300 K and an environment temperature of T_0 = 298 K, the reversible thermal efficiency found with (5.9) is 77.1%. Dividing the actual thermal efficiency of 34.1% by this efficiency (0.341/0.771) gives 44.2%. Note that this is the same as the exergy efficiency obtained using (5.7).

The results of the numerical example considered in this section are shown in a combined energy and exergy diagram in Fig. 5.2. In many studies with energy and exergy analyses of power cycles, energy and exergy flow diagrams are given separately. The combined flow diagram approach used here appears to be useful in conveying energy and exergy results of the cycle in a scaled, compact, and comprehensive manner. The heating value of the fuel is normalized to 100 units of energy and other values are normalized accordingly. The thermal and exergy efficiencies discussed in this section can easily be obtained using the values in this diagram by taking the ratios of various terms. The total exergy destruction in this power plant is 78 kJ for a total exergy input of 103.6 kJ. The exergy destruction in the cycle based on an exergy input of 33.2 kJ is only 7.6 kJ (33.2–25.6), which is only 9.7% of total exergy destruction. That is, the exergy destructions in the furnace–boiler system account for the remaining 90.3% of the total exergy destruction. This significant exergy destruction is not considered in an exergy efficiency definition neglecting the destructions in the furnace–boiler system [see (5.4)].

One may question the value of exergy analysis as a tool for assessing a power plant because the thermal efficiency based on the heating value of the fuel [(5.3)] and the exergy efficiency based on the exergy of the fuel [(5.8)] are very close. Although the exergy efficiency in this case adds little new information for addressing cycle efficiency, we have to remember that a major use of exergy analysis is to analyze the system components separately and to identify and quantify the sites of exergy destruction. This information can then be used to improve the performance of the system by trying to minimize the exergy destructions in a prioritized manner. Note that the exergy efficiency defined in (5.6) addresses the fact that only a fraction of the heat from combustion that is transferred to the steam in the boiler is available for work, and the exergy efficiency compares the actual work output to this available work (i.e., exergy). The exergy efficiencies in these cases become greater than the corresponding thermal efficiencies, providing more realistic measures of system performance compared to the corresponding thermal efficiencies. For a more comprehensive thermodynamic analysis of a power cycle, the various energy- and exergy-based efficiencies are best considered.

5.3 Efficiencies of Gas Power Plants

The schematic of an open-cycle gas turbine power plant is given in Fig. 5.3. The thermal efficiency of this plant may be expressed as

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{fuel}} q_{\text{HV}}} \tag{5.10}$$

The exergy efficiency of this gas-turbine engine is

$$\varepsilon = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{fuel}} x_{\text{fuel}}} \tag{5.11}$$

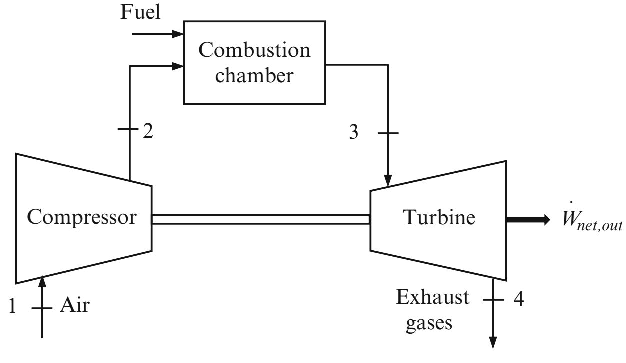

Fig. 5.3 An open-cycle gas-turbine engine

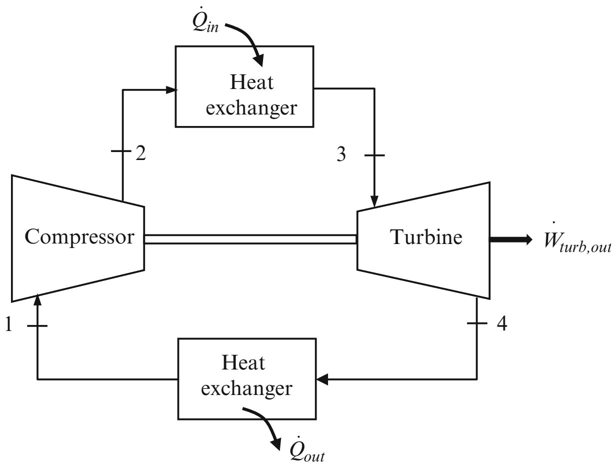

Fig. 5.4 A closed-cycle gas-turbine engine

This engine is sometimes modeled by a closed-cycle gas-turbine engine as shown in Fig. 5.4. The working fluid is assumed to be air and the combustion process is replaced by a heat addition process. In this cycle, the energy efficiency may be written as

$$\eta_{\mathrm{th}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_{\mathrm{fuel}} q_{\mathrm{HV}}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_{\mathrm{air}} q_{\mathrm{in}}} = \frac{\dot{W}_{\mathrm{turb,\ out}} - \dot{W}_{\mathrm{comp,\ in}}}{\dot{m}_{\mathrm{air}}(h_3 - h_2)} \tag{5.12}$$

Note that the heat added to the cycle is equal to the heat resulting from the combustion process. Equation 5.12 is equivalent to (5.10). The exergy efficiency may be written by different approaches as

$$\varepsilon = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{Q}_{\mathrm{in}}\left(1 - \frac{T_0}{T_s}\right)} \tag{5.13}$$

$$\varepsilon = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{X}_3 - \dot{X}_2} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_{\mathrm{air}}[h_3 - h_2 - T_0(s_3 - s_2)]} \tag{5.14}$$

Here, (5.11, 5.13), and (5.14) give different results for the exergetic efficiency values. Equation 5.13 does not account for the exergy destruction during the combustion process whereas (5.14) does not account for the exergy destructions during combustion and during the heat transfer to the working fluid in the cycle. The efficiency will be highest in (5.14) and lowest in (5.11).

Energy and exergy efficiencies of an actual internal combustion engine can be expressed using (5.10) and (5.11). For the idealized models of internal combustion engines (Otto, Diesel, and Dual cycles), modified versions of (5.12) through (5.14) may easily be obtained using the same principles.

Simplified thermal efficiency relations of idealized cycles for internal combustion engines and gas-turbine cycles are available. When the Otto cycle is used to represent the operation of an internal combustion engine, the thermal efficiency under air-standard assumptions (working fluid is air; air is an ideal gas with constant specific heats) is

$$\eta_{\text{th, Otto}} = 1 - \frac{1}{r^{k-1}} \tag{5.15}$$

where r is the compression ratio and k is the specific heat ratio. Similarly, the thermal efficiency of the Diesel cycle, which is the idealized model for compression ignition engines, is

$$\eta_{\text{th, Diesel}} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_c^k - 1}{k(r_c - 1)} \right] \tag{5.16}$$

where r_c is the cutoff ratio, defined as the ratio of cylinder volumes after and before the combustion process. The efficiency relation for the Dual cycle is

$$\eta_{\text{th, Dual}} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_p r_c^k - 1}{k r_p (r_c - 1) + r_p - 1} \right] \tag{5.17}$$

where r_p is the ratio of pressures after and before the constant-volume heat addition process. The thermal efficiency of the simple Brayton cycle, which is the idealized model for gas-turbine engines, is expressed using the air-standard assumption as

$$\eta_{\text{th, Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}} \tag{5.18}$$

where r_p is the ratio of maximum and minimum pressures in the cycle. For the idealized regenerative Brayton cycle, the efficiency relation is

$$\eta_{\text{th, Brayton, regen}} = 1 - \left(\frac{T_1}{T_3} \right) r_p^{(k-1)/k} \tag{5.19}$$

where T_1 and T_3 are the temperatures at the inlets of the compressor and the turbine, respectively.

The operational description of these idealized cycles may be found in most thermodynamics textbooks [1, 23]. Equations 5.15 through 5.19 are only applicable to the idealized cycles considered, and they should not be used to determine the thermal efficiencies of actual internal combustion engines or gas-turbine cycles. Equations 5.15 through 5.19 are useful in that they illustrate the effects of some key design parameters such as compression ratio, cutoff ratio, and pressure ratio on cycle efficiency.

5.4 Efficiencies of Cogeneration Plants

Cogeneration refers to the simultaneous generation of more than one form of energy product. Performance assessment of various cogeneration plants are given in Kanoglu and Dincer [24]. For a cogeneration plant producing electric power $\dot{W}_{\text{net,out}}$ and process heating $\dot{Q}_{\text{process}}$, a first-law-based efficiency is defined as the ratio of useful energy output to energy input:

$$\eta_{\text{cogen}} = \frac{\dot{W}_{\text{net, out}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = 1 - \frac{\dot{Q}_{\text{loss}}}{\dot{Q}_{\text{in}}} \tag{5.20}$$

where $\dot{Q}_{\text{process}}$ is the output rate of process heat and $\dot{Q}_{loss}$ is the heat lost in the condenser. This relation is referred to as the utilization efficiency to differentiate it from the thermal efficiency which is used for a power plant where the single output is power. Students are consistently taught not to compare apples and oranges, which usually refers to two commodities that are different. Work and heat have the same units but are fundamentally difficult to add because they are different, with work being a more valuable commodity than heat.

We can overcome this situation by defining the efficiency of a cogeneration plant based on exergy, as the ratio of total exergy output to exergy input:

$$\varepsilon_{\text{cogen}} = \frac{\dot{X}_{\text{out}}}{\dot{X}_{\text{in}}} = \frac{\dot{W}_{\text{net, out}} + \dot{X}_{\text{process}}}{\dot{X}_{\text{in}}} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_{\text{in}}} \tag{5.21}$$

where $\dot{X}_{\text{process}}$ is the exergy transfer rate associated with the transfer of process heat, expressible as

$$\dot{X}_{\text{process}} = \int \delta\dot{Q}_{\text{process}}\left(1 - \frac{T_0}{T}\right) \tag{5.22}$$

where T is the instantaneous source temperature from which the process heat is transferred. This relation is of little practical value unless the functional relationship between the process heat rate $\dot{Q}_{\text{process}}$ and temperature T is known. In many cases, the process heat is utilized by the transfer of heat from a working fluid exiting the heat producing device (e.g., a turbine or an internal combustion engine) to a secondary fluid in a heat exchanger (Fig. 5.5). One can express the exergy rate of process heat as the exergy decrease of the hot fluid in the heat exchanger as

$$\dot{X}_{\text{process}-1} = -\Delta\dot{X}_{\text{hot}} = \dot{m}_{\text{hot}}[h_1 - h_2 - T_0(s_1 - s_2)]_{\text{hot}} \tag{5.23}$$

or by the increase of the exergy of the cold fluid in the heat exchanger

$$\dot{X}_{\text{process}-2} = \Delta\dot{X}_{\text{cold}} = \dot{m}_{\text{cold}}[h_4 - h_3 - T_0(s_4 - s_3)]_{\text{cold}} \tag{5.24}$$

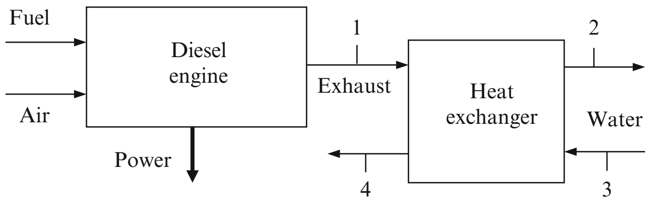

Fig. 5.5 Cogeneration plant with a diesel engine and a heat exchanger for steam production

where the subscripts refer to state points in Fig. 5.5. The difference between these two exergies is the exergy destruction in the heat exchanger. Then, from (5.21), the exergy efficiencies based on these two approaches become

$$\varepsilon_{\text{cogen}-1} = \frac{\dot{W}_{\text{net, out}} + \dot{m}_{\text{hot}}[h_1 - h_2 - T_0(s_1 - s_2)]_{\text{hot}}}{\dot{X}_{\text{in}}} \tag{5.25}$$

and

$$\varepsilon_{\text{cogen}-2} = \frac{\dot{W}_{\text{net, out}} + \dot{m}_{\text{cold}}[h_4 - h_3 - T_0(s_4 - s_3)]_{\text{cold}}}{\dot{X}_{\text{in}}} \tag{5.26}$$

The exergy input in these relations can be expressed differently using various inputs as in the denominators of (5.6) through (5.8), yielding different exergy efficiencies.

Example 5.2 To illustrate the use of these efficiencies, we consider a diesel engine-based cogeneration plant, The outputs are electrical power and process heat, which is transferred from the hot exhaust gases to water to produce steam in a heat exchanger (Fig. 5.5). Some of the data used in this example are from an actual diesel engine power plant [25]. The net power output from the plant is 18,900 kW when the rate of fuel consumption rate is 1.03 kg/s and the air–fuel ratio is 40.4. This corresponds to an exhaust flow rate of 41.6 kg/s. The plant uses heavy diesel fuel with a lower heating value of 39,300 kJ/kg. The exhaust gases enter the process heating unit (i.e., heat exchanger) at 383°C and experience a temperature drop of 175°C whereas compressed liquid water enters at 15°C and exits as saturated vapor at 200°C. Applications of (5.20) through (5.26) produce the following results.

$$\dot{Q}_{\text{process}} = 7784\,\text{kW},\ \dot{X}_{\text{in}} = 43,110\,\text{kW},\ \dot{X}_{\text{process}-1} = 3678\,\text{kW}$$

$$\dot{X}_{\text{process}-2} = 2509\,\text{kW},\ \eta_{\text{cogen}} = 65.9\%,\ \varepsilon_{\text{cogen}-1} = 52.4\%,\ \varepsilon_{\text{cogen}-2} = 49.7\%$$

The exergy of heavy diesel fuel with an unknown composition is taken as 1.065 times the lower heating value of the fuel following the approach by Brzustowski and Brena [26]. Properties of air with variable specific heats are used for exhaust gases.

The difference between the energy and exergy efficiencies in this cogeneration plant appears to be much greater than the difference for a power plant, when the

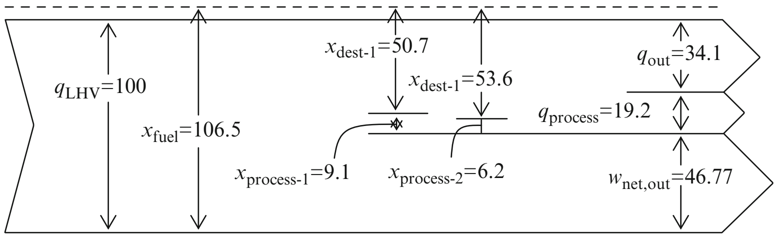

Fig. 5.6 Combined energy and exergy diagram for the cogeneration plant considered

energy and exergy efficiencies are respectively defined based on the energy and exergy of the fuel, as discussed in the previous section. The difference is attributable to one of the product outputs being process heat. The different approaches used in (5.18) and (5.19) to define the exergy of the process heat result in a small exergy efficiency difference of only $52.4-49.7 = 2.7\%$. The greater the average temperature difference between the hot and cold fluids in the process heater, the greater is the exergy destruction and the greater is the difference between the two definitions of exergy efficiencies in (5.25) and (5.26), respectively.

The results are presented in a combined energy and exergy diagram in Fig. 5.6. The heating value of the fuel (i.e., heat input) is normalized to 100 units of energy and other values are modified accordingly. The energy and exergy efficiencies discussed in this section can be found using this diagram. The total exergy destruction in this cogeneration plant is 50.7 kJ based on the first approach and 53.6 kJ based on the second approach, for a total exergy input of 106.5 kJ. The difference between these exergy destructions is the exergy destruction in the process heater, which is 5.7% of total exergy destruction or 2.7% of exergy input.

5.4.1 Steam-Turbine-Based Cogeneration Plant

Referring to Fig. 5.7 for the states, the energy efficiency is expressible as

$$\eta_{\text{cogen}} = \frac{\dot{W}_{\text{net}} + \dot{m}_{water}(h_{10} - h_9)}{\dot{m}_{\text{fuel}} q_{\text{LHV}}} \tag{5.27}$$

where $\dot{m}_{\text{fuel}}$ is the mass flow rate of fuel and q_{LHV} is the lower heating value of the fuel. Higher heating value can also be used in this equation. Using a higher heating value would correspond to a lower energy efficiency compared to using a lower heating value.

The value of the mass flow rate of the steam extracted from the turbine at state 5 (Fig. 5.7) affects the energy efficiency of the cogeneration plant. The greater the amount of mass extracted is, the greater the amount of heating and the smaller the

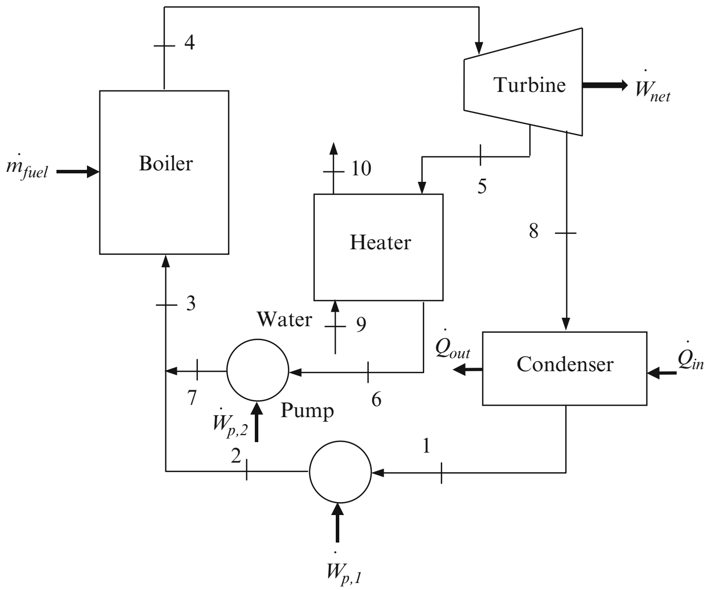

Fig. 5.7 A steam-turbine cogeneration plant

power output. The optimum value should be selected depending on the heat and power demands.

The exergy input to a steam cogeneration plant can be expressed as

$$\dot{X}_{\text{in}} = \dot{m}_{\text{fuel}} x_{\text{fuel}} \tag{5.28}$$

where x_{fuel} is the specific exergy of the fuel. The exergy of a fuel may be obtained by writing the complete combustion reaction of the fuel and calculating the reversible work obtainable assuming all products are at the state of surroundings. The exergy of the fuel is equal to the reversible work. For fuels that yield water as a combustion product, the exergy of the fuel differs depending on the water's phase (vapor or liquid). Szargut et al. [10] list the exergies of various fuels based on the vapor phase of water in combustion gases.

Referring to the states as given in Fig. 5.7, we obtain the exergy efficiency as

$$\varepsilon_{\text{cogen}} = \frac{\dot{W}_{\text{net}} + \dot{m}_{\text{water}}[h_{10} - h_9 - T_0(s_{10} - s_9)]}{\dot{m}_{\text{fuel}} x_{\text{fuel}}} \tag{5.29}$$

As an illustrative example, we consider a cogeneration steam power plant with a net power output of 10 MW and boiler and condenser pressures of 10,000 and 10 kPa, respectively (Fig. 5.7). The turbine inlet temperature is 500°C and isentropic efficiencies for both the turbine and the pump are assumed to be 85%. Steam is extracted from the turbine at 2 MPa pressure and used to obtain hot water for the radiator from the heater. The steam exits the heater at the same pressure as a saturated liquid. The liquid water, heated to 90°C, is used to heat buildings and returns to the cogeneration plant at 50°C as a common practice.

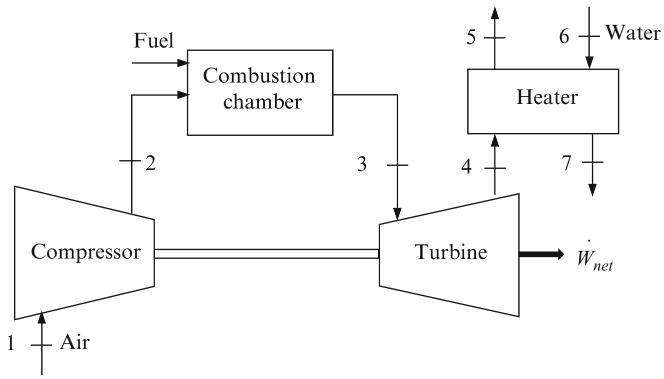

Fig. 5.8 A gas-turbine cogeneration plant

The energy efficiency of the furnace–boiler system is taken to be 90%. That is, 90% of the lower heating value of the fuel is transferred to the steam generated in the boiler and the remaining 10% is lost, mostly with the hot exhaust gases passing through the chimney. The dead-state temperature is taken to be 25°C. We consider methane as the fuel with a lower heating value of 50,050 kJ/kg and a chemical exergy of 51,840 kJ/kg [10].

For a net power output of 10 MW, the mass flow rate through the boiler is 16.17 kg/s and the extracted steam mass flow from the turbine for heating water is 6.78 kg/s. Under these operating conditions, the mass flow rate of liquid water is 80.5 kg/s and the corresponding heat transfer in the heater is 13.5 MW.

5.4.2 *Gas-Turbine-Based Cogeneration Plant*

The energy efficiency of a gas-turbine cogeneration plant can be expressed as

$$\eta_{\mathrm{cogen}} = \frac{\dot{W}_{\mathrm{net}} + \dot{m}_{\mathrm{water}}(h_7 - h_6)}{\dot{m}_{\mathrm{fuel}} q_{\mathrm{LHV}}} \tag{5.30}$$

where $\dot{m}_{\mathrm{water}}$ is the mass flow rate of water and the state numbers are shown in Fig. 5.8. The exergy efficiency is given by

$$\varepsilon_{\mathrm{cogen}} = \frac{\dot{W}_{\mathrm{net}} + \dot{m}_{\mathrm{water}}[h_7 - h_6 - T_0(s_7 - s_6)]}{\dot{m}_{\mathrm{fuel}} x_{\mathrm{fuel}}} \tag{5.31}$$

As an illustrative example, we consider a cogeneration gas-turbine power plant with a net power output of 10 MW and maximum and minimum pressures of 1,200 and 100 kPa in the system, respectively (Fig. 5.8). The fuel is methane, the turbine inlet temperature is 700°C and the isentropic efficiencies of both the turbine and compressor are 85%. Under these operating conditions, exhaust gases leave the turbine at 303°C

and the mass flow rate steam through the turbine is 141.9 kg/s. For the water heated in the heater, the same inlet and exit temperatures and mass flow rate as in Sect. 5.3 are considered, so the heat transfer rate in the heater remains 13.5 MW.

5.5 Efficiencies of Geothermal Power Plants

The technology for producing power from geothermal resources is well established and there are many geothermal power plants operating worldwide [27]. Depending on the state of the geothermal fluid in the reservoir, different power-producing cycles may be used including direct steam, flash-steam (single and double-flash), binary and combined flash–binary cycles. In general, the thermal efficiency of a geothermal power plant may be expressed as

$$\eta_{\mathrm{th}-1} = \frac{\dot{W}_{\mathrm{net,\,out}}}{\dot{E}_{\mathrm{in}}} \tag{5.32}$$

where $\dot{E}_{\mathrm{in}}$ is the energy input rate to the power plant, which may be expressed as the specific enthalpy of the geothermal water with respect to the environment state multiplied by the mass flow rate of geothermal water $\dot{m}_{\mathrm{geo}}$. That is,

$$\eta_{\mathrm{th}-1} = \frac{\dot{W}_{\mathrm{net,\,out}}}{\dot{m}_{\mathrm{geo}}(h_{\mathrm{geo}} - h_0)} \tag{5.33}$$

The state of the geothermal water may be taken as that in the reservoir or at the well head. Those who use the reservoir state argue that a realistic and more meaningful comparison between geothermal power plants needs to account for methods of harvesting the geothermal fluid. However, those who use the well-head-state argue that taking the reservoir as the input is not appropriate for geothermal power plants because conventional power plants are evaluated on the basis of the energy of the fuel burned at the plant site [28–30]. In (5.33), the energy input to the power plant represents the maximum heat the geothermal water can deliver, which occurs when the geothermal water is cooled to the temperature of the environment.

The simplest geothermal cycle is the direct steam cycle. Steam from the geothermal well is passed through a turbine and exhausted to the atmosphere or to a condenser. Flash steam plants are used to generate power from liquid-dominated resources that are hot enough to flash a significant proportion of the water to steam in surface equipment, either at one or two pressure stages (single-flash or double-flash plants) as shown in Figs. 5.9 and 5.10, respectively. The steam flows through a steam turbine to produce power while the brine is reinjected back to the ground. Steam exiting the turbine is condensed with cooling water obtained in a cooling tower or a spray pond before being reinjected. Binary cycle plants use the geothermal brine from liquid-dominated resources (Fig. 5.11). These plants operate on a Rankine cycle with a binary

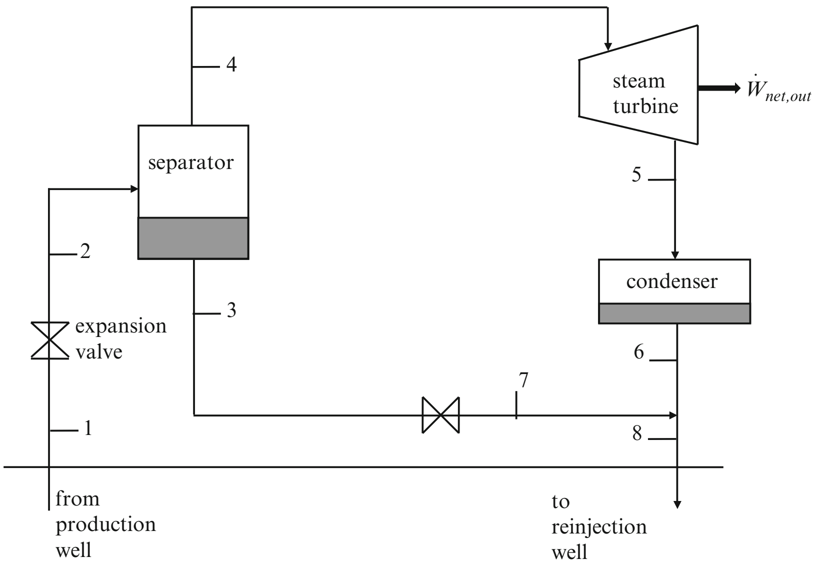

Fig. 5.9 Single-flash geothermal power plant

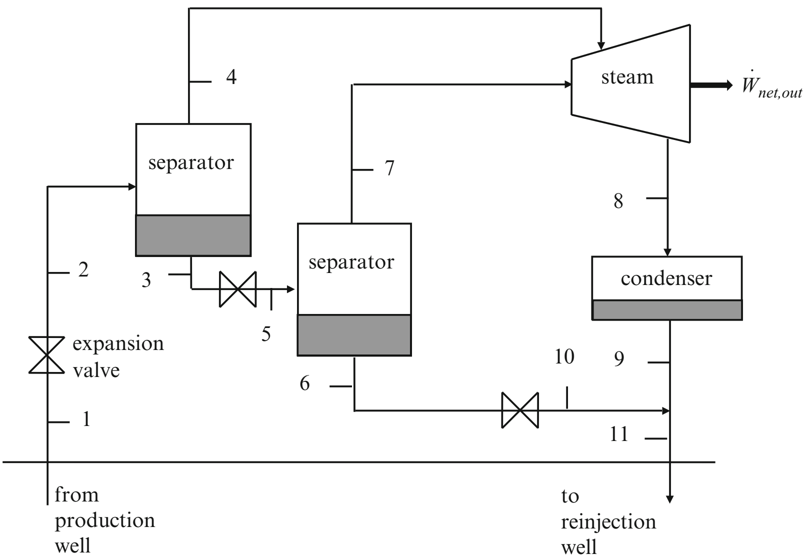

Fig. 5.10 Double-flash geothermal power plant

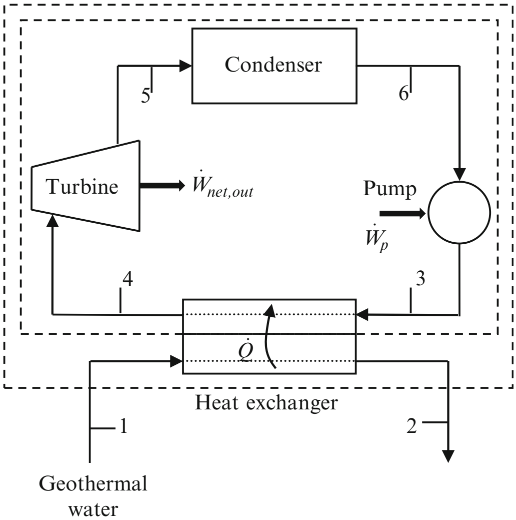

Fig. 5.11 Binary geothermal power plant

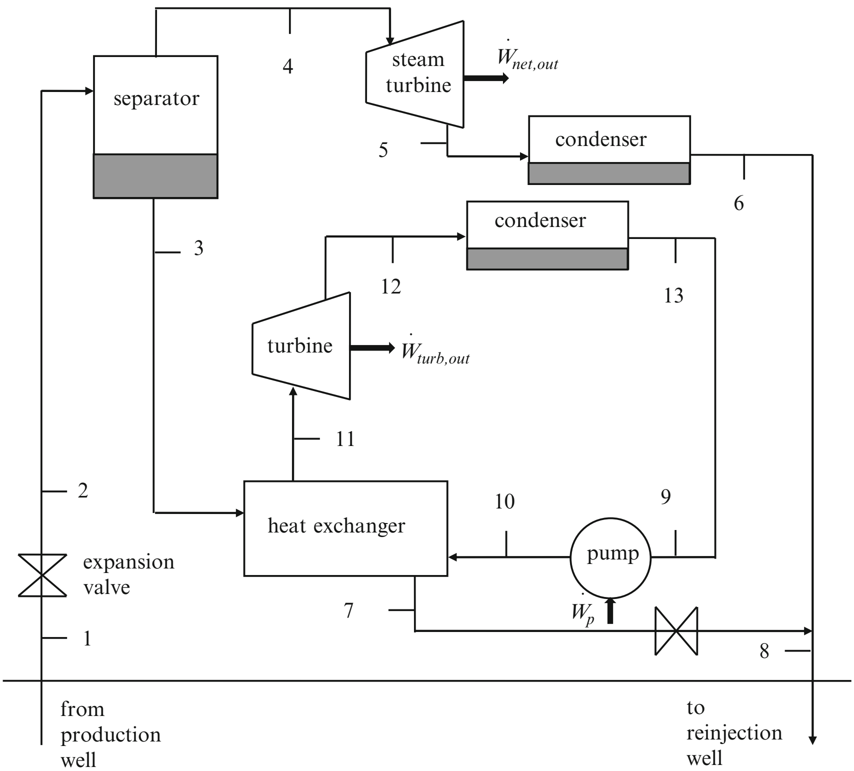

Fig. 5.12 Combined flash-binary geothermal power plant

working fluid (isobutane, isopentane, R-114, etc.) that has a low boiling temperature. The working fluid is completely vaporized and usually superheated by the geothermal heat in the vaporizer. The vapor expands in the turbine, and then condenses in a water-cooled condenser or dry cooling tower before being pumped back to the vaporizer to complete the cycle. Combined flash/binary plants (Fig. 5.12) incorporate both a binary unit and a flashing unit to exploit the advantages associated with both systems. The liquid portion of the geothermal mixture serves as the input heat for the binary cycle and the steam portion drives a steam turbine to produce power.

The actual heat input to a geothermal power cycle is less than the term in the denominator of (5.33) inasmuch as part of geothermal water is reinjected back to the ground at a temperature much greater than the temperature of the environment. In an approach that accounts for the actual reinjection temperature, the thermal efficiency is expressed as

$$\eta_{\mathrm{th-2}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{Q}_{\mathrm{in}}} \tag{5.34}$$

For a single-flash cycle, the thermal efficiency may be expressed as

$$\eta_{\mathrm{th,\ single\ flash}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{Q}_{\mathrm{in}}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_2 h_2 - \dot{m}_3 h_3} \tag{5.35}$$

where the subscripts refer to state points in Fig. 5.9. For a double-flash cycle, the efficiency becomes

$$\eta_{\mathrm{th,\ double\ flash}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{(\dot{m}_2 h_2 - \dot{m}_3 h_3) + (\dot{m}_5 h_5 - \dot{m}_6 h_6)} \tag{5.36}$$

where the state points are shown in Fig. 5.10. Referring to Fig. 5.11, for a binary cycle we obtain

$$\eta_{\mathrm{th,\ binary}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_{\mathrm{geo}}(h_1 - h_2)} \tag{5.37}$$

or

$$\eta_{\mathrm{th,\ binary}} = \frac{\dot{W}_{\mathrm{net,\ out}}}{\dot{m}_{\mathrm{binary}}(h_4 - h_3)} \tag{5.38}$$

where $\dot{m}_{\mathrm{binary}}$ is the mass flow rate of binary working fluid. For a combined flash-binary cycle, the thermal efficiency is

$$\eta_{\text{th, flash-binary}} = \frac{\dot{W}_{\text{net, out}}}{(\dot{m}_2 h_2 - \dot{m}_3 h_3) + (\dot{m}_3 h_3 - \dot{m}_7 h_7)} \tag{5.39}$$

where the state points are shown in Fig. 5.12.

Using the exergy of geothermal water (in the reservoir or at the well head) as the exergy input to the plant, the exergy efficiency of a geothermal power plant can be expressed as

$$\varepsilon = \frac{\dot{W}_{\text{net, out}}}{\dot{X}_{\text{in}}} = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{geo}}\left[h_{\text{geo}} - h_0 - T_0\left(s_{\text{geo}} - s_0\right)\right]} \tag{5.40}$$

Using the exergy change of geothermal water in the cycle as the exergy input to the cycle, the exergy efficiencies may be expressed for single-flash, double-flash, and combined flash–binary cycles as

$$\begin{aligned}\varepsilon_{\text{single flash}} &= \frac{\dot{W}_{\text{net, out}}}{\dot{m}_2 x_2 - \dot{m}_3 x_3} \\ &= \frac{\dot{W}_{\text{net, out}}}{\dot{m}_2[h_2 - h_0 - T_0(s_2 - s_0)] - \dot{m}_3[h_3 - h_0 - T_0(s_3 - s_0)]}\end{aligned} \tag{5.41}$$

$$\varepsilon_{\text{double flash}} = \frac{\dot{W}_{\text{net, out}}}{(\dot{m}_2 x_2 - \dot{m}_3 x_3) + (\dot{m}_5 x_5 - \dot{m}_6 x_6)} \tag{5.42}$$

$$\varepsilon_{\text{flash-binary-1}} = \frac{\dot{W}_{\text{net, out}}}{(\dot{m}_2 x_2 - \dot{m}_3 x_3) + (\dot{m}_3 x_3 - \dot{m}_7 x_7)} \tag{5.43}$$

where *ex* is the specific flow exergy of the fluid. For a binary cycle, the exergy efficiency may be defined based on the exergy decrease of geothermal water or the exergy increase of the binary working fluid in the heat exchanger. That is,

$$\varepsilon_{\text{binary-1}} = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{geo}}[h_2 - h_1 - T_0(s_2 - s_1)]} \tag{5.44}$$

$$\varepsilon_{\text{binary-2}} = \frac{\dot{W}_{\text{net, out}}}{\dot{m}_{\text{binary}}[h_4 - h_3 - T_0(s_4 - s_3)]} \tag{5.45}$$

The difference between the denominators of (5.44) and (5.45) is the exergy destruction in the heat exchanger. The exergy efficiency definitions in (5.44) and (5.45) can be illustrated by considering the different systems indicated by the inner and outer dashed lines, respectively, in Fig. 5.11.

By adapting the approach used in (5.45), one may express the exergy efficiency for a combined flash–binary cycle as

$$\varepsilon_{\text{flash-binary-2}} = \frac{\dot{W}_{\text{net, out}}}{(\dot{m}_2 x_2 - \dot{m}_3 x_3) + \dot{m}_{\text{binary}}(x_{11} - x_{10})} \tag{5.46}$$

The efficiency in (5.43) is more advantageous than that in (5.46) because exergy input is expressed by the exergy change of geothermal water for both the flash and binary parts of the cycle in (5.43), respectively.

Example 5.3 Consider a binary geothermal power plant like that in Fig. 5.11 using geothermal water at 165°C with isobutane as the working fluid. The mass flow rate of geothermal water is 555 kg/s. In this cycle, isobutane is heated and vaporized in the heat exchanger by geothermal water. Then the isobutane flows through the turbine, is condensed, and pumped back to the heat exchanger, completing the binary cycle. The heat exchanger and condenser pressures are taken to be 3,000 and 400 kPa, respectively, and the temperature at the turbine inlet (or heat exchanger exit) is taken to be 150°C, which is 15°C lower than the geothermal water temperature at the heat exchanger inlet. The isentropic efficiencies of the turbine and pump are taken to be 80% and 70%, respectively. About 10% of the power output is used for internal demands such as powering fans in the air-cooled condenser. These values closely correspond to those of an actual power plant [31]. Noting that a pinch-point will occur at the start of vaporization of the working fluid in the heat exchanger, the energy balance relations for the heat exchanger can be written as

$$\dot{m}_{\text{geo}} c_{\text{geo}} \left[T_1 - \left(T_{\text{vap}} + \Delta T_{\text{pp}}\right)\right] = \dot{m}_{\text{binary}} \left(h_4 - h_{\text{binary},f}\right) \tag{5.47}$$

$$\dot{m}_{\text{geo}} c_{\text{geo}} \left[\left(T_{\text{vap}} + \Delta T_{\text{pp}}\right) - T_2\right] = \dot{m}_{\text{binary}} \left(h_{\text{binary},f} - h_3\right) \tag{5.48}$$

where $\dot{m}_{\text{binary}}$ is the mass flow rate of the binary fluid, c_{geo} is the specific heat of geothermal water, T_{vap} is the vaporization temperature of the binary fluid at the heat exchanger pressure, ΔT_{pp} is the pinch-point temperature difference, and $h_{\text{binary},f}$ is the specific enthalpy of the binary fluid at the start of vaporization. The pinch-point temperature difference is assumed to be 6°C. Equations 5.47 and 5.48 can be used to establish the mass flow rate of the binary fluid, and the geothermal water temperature at the heat exchanger exit. The analysis of the cycle with the stated values produces the following results.

$$\dot{W}_{\text{net, out}} = 22{,}382\,\text{kW},\ \dot{E}_{\text{in}} = 328{,}786\,\text{kW},\ \dot{Q}_{\text{in}} = 185{,}181\,\text{kW},\ T_2 = 86.6°\text{C}$$

$$\dot{X}_{\text{in}} = 60{,}014\,\text{kW},\ \Delta\dot{X}_{1-2} = 46{,}904\,\text{kW},\ \Delta\dot{X}_{3-4} = 37{,}316\,\text{kW},$$

$$n_{\text{th}-1} = 6.8\%\ [\text{Eq.}(5.33)],\ \eta_{\text{th}-2} = 12.1\%\ [\text{Eq.}(5.37)]$$

$$\varepsilon = 37.3\%\ [\text{Eq.}(5.40)],\ \varepsilon_{\text{binary}-1} = 47.7\%\ [\text{Eq.}(5.44)],$$
$$\varepsilon_{\text{binary}-2} = 60.0\%\ [\text{Eq.}(5.45)]$$

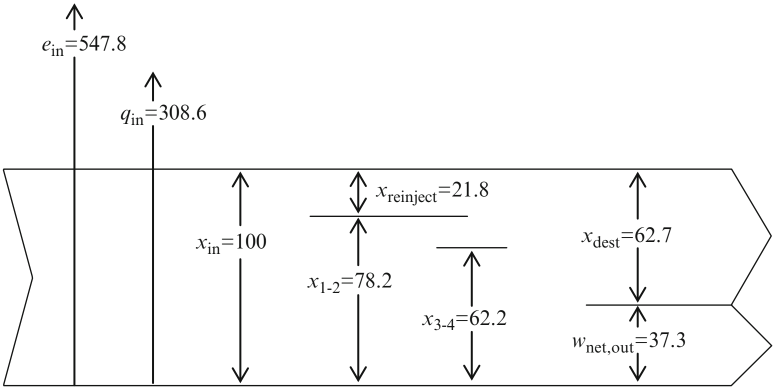

Fig. 5.13 Combined energy and exergy diagram for the binary geothermal power plant considered

It is clear that using different definitions leads to significantly different thermal and exergy efficiencies. This is typical of geothermal power plants. The results are presented in a combined energy and exergy diagram in Fig. 5.13. Because of the large range of values involved, the exergy of geothermal water at the heat exchanger inlet is normalized to 100 units of energy and other values are modified accordingly. The energy and exergy efficiencies can be obtained using terms in this diagram. The thermal and exergy efficiencies are 6.8% and 37.3%, respectively, based on the energy and exergy of geothermal water at the heat exchanger inlet. The thermal efficiency increases from 6.8% to 12.1% when the heat input to the binary fluid in the heat exchanger is used as the energy input to the cycle instead of the energy of the geothermal water at the inlet of the plant. This is analogous to using the heating value of fuel versus using the heat transferred to the steam in the boiler as the heat input to a steam power plant.

The exergy efficiency is only 37.3% when the exergy of geothermal water at the plant inlet is used. Using the exergy decrease of geothermal water in the heat exchanger as the exergy input to the cycle yields an exergy efficiency of 47.7% whereas using the exergy increase of the binary fluid yields an exergy efficiency of 60.0%. These three approaches are analogous to using the exergy of the fuel [(5.8)], the exergy transfer to the steam accompanying the heat input to the cycle [(5.7)], and the exergy increase of the steam in the boiler [(5.6)]. The exergy of the geothermal water at the exit of the heat exchanger, which is reinjected to the ground, represents 21.8% of the exergy input to the cycle. This significant percentage is due to the relatively high temperature of the geothermal water (86.6°C). The exergy destruction in the heat exchanger accounts for 16.0% of the exergy input. The remaining exergy destructions ($100 - 21.8 - 16.0 - 37.3 = 24.9$) are due to irreversibilities in the turbine, pump, and condenser.

The efficiencies for the plants considered as examples yield some important information on the relative magnitudes of heat losses and exergy destructions in the plants. Combined energy and exergy diagrams present the results concisely and

clearly. The efficiencies for power cycles not specifically discussed in this chapter can be deduced from the relations given for the cycles considered.

For the current state of thermodynamics, it seems almost impossible to have a common efficiency definition for all energy systems. Therefore, the best way of avoiding misuse and misunderstanding is to define the efficiency used in any application carefully. An understanding of both energy and exergy efficiencies is essential for designing, analyzing, optimizing, and improving energy systems through appropriate energy policies and strategies. If such policies and strategies are in place, numerous measures can be applied to improve the efficiency of electrical generating plants. These measures should be weighed against other factors and, where appropriate, implemented. It should be understood that decisions on power plant operations are normally based primarily on economic criteria. Often other criteria such as environmental considerations are also important. Economic and exergy analyses can be combined by means of exergoeconomic analyses, which can include exergetic life cycle assessment. A rational efficiency definition should accompany such an analysis. It is more appropriate to use an exergy efficiency based on the exergy of the fuel in a fossil-fuel power or cogeneration system because an important part of exergy costing involves fuel cost. All exergy losses are accounted for in this approach. For renewable energy systems, it is more appropriate to use the exergy of an energy source as the exergy input to the system. This approach allows all exergy destructions to be accounted for, including those in heat exchange equipment. All losses are ultimately related to the economics of the system operation. The difference between efficiency definitions often relates to the selection of different system boundaries. Depending on the selection, losses occurring at a particular site may be accounted for in a definition or excluded [6].

A detailed case study on efficiency evaluation of a binary geothermal power plant is given in Appendix A.

5.6 Energetic and Exergetic Analyses of a Photovoltaic System

This analysis is based on references [32–34]. An example of solar technology is adopted to demonstrate the link between sustainability and efficiency. An effective way to maintain a good electrical efficiency by removing heat from the solar panels and to have a better overall efficiency of a photovoltaic system is to utilize both technologies simultaneously. This kind of system is known as a hybrid photovoltaic/thermal (PV/T) system and can be beneficial for low-temperature thermal applications such as water heating, air heating, agricultural crop drying, solar greenhouses, and space heating among others, along with electricity generation that can further be beneficial for rural electrification and agricultural applications including solar water pumping and so on. In this case study we give a simple demonstration of how both technologies together give better efficiency, which directly relates to better sustainability.

Based on the first law of thermodynamics, the energy efficiency of a PV/T system can be defined as a ratio of total energy (electrical and thermal) produced by a PV/T system to the total solar energy falling on the photovoltaic surface and can be given as [19, 20]:

$$\eta = \frac{E}{S_T A} = \frac{V_{oc} I_{sc} + \dot{Q}}{S_T A} \tag{5.49}$$

where

$$\dot{Q} = h_{ca} A (T_{cell} - T_{amb}) \text{ and } h_{ca} = 5.7 + 3.8v.$$

Here, h_{ca}, A, T_{cell}, T_{amb}, I_{sc}, and V_{oc} are the convective heat transfer coefficient from the photovoltaic cell to the ambient area of the photovoltaic surface, cell temperature, ambient temperature, short circuit current, and open circuit voltage, respectively. The convective (and radiative) heat transfer coefficient from the photovoltaic cell to ambient, can be calculated by considering wind velocity (v), density of the air, and the surrounding (ambient) conditions.

The exergy efficiency is based on the second law of thermodynamics that not only gives the quantitative assessment of energy but also the qualitative. This is also called exergy efficiency. A comparison of the PV and PV/T systems is also presented in the form of a case study later in this section.

The exergy efficiency of a photovoltaic system can be given as

$$\varepsilon = \frac{\dot{X}}{\dot{X}_{solar}} \tag{5.50}$$

where $\dot{X}$ is the exergy of the PV system which is mainly the electrical power output of the system. The thermal energy gained by the system during operation is not desirable in the case of a PV system, therefore this becomes a heat loss to the system and hence needs to be subtracted from the former in order to calculate the exergy of a PV system. $\dot{X}_{solar}$ is the exergy rate from the solar irradiance in W/m^2 which can be given as

$$\dot{X}_{solar} = \left(1 - \frac{T_{amb}}{T_{sun}}\right) S_T A \tag{5.51}$$

An expression for the exergy of PV can be given as

$$\dot{X} = V_m I_m - \left(1 - \frac{T_{amb}}{T_{cell}}\right) \dot{Q} \tag{5.52}$$

Here, I_m and V_m are the actual current and voltage.

Unlike PV systems, the PV/T system uses the thermal energy available on the PV panel and this time the thermal energy gain can be utilized as useful energy and hence, the exergy of the PV/T system becomes the sum of the electrical exergy and thermal exergy of the system and the exergy efficiency can be defined as

$$\varepsilon = \frac{\dot{X}}{\dot{X}_{solar}} = \frac{\dot{X}_e + \dot{X}_{th}}{\dot{X}_{solar}} \tag{5.53}$$

An expression for the exergy of the PV/T system can be given as [20]:

$$\dot{X} = V_m I_m + \left(1 - \frac{T_{amb}}{T_{cell}}\right)\dot{Q} \tag{5.54}$$

We now apply the model presented above to some actual data sets as obtained through experiments in New Delhi, India, which is located at 77°12′E longitude and 28°35′N latitude. The test was performed from 9:00 a.m. to 4:00 p.m. on March 27, 2006 and the data measured included total solar irradiation, voltage, open-circuit voltage, current, short-circuit current, cell temperature, ambient temperature, and velocity of the air just above the photovoltaic surface. The data for hourly total solar radiation and the wind velocity were measured for different places on the photovoltaic surface and an average value for both was used to calculate the energy and exergy of the photovoltaic system. The uncertainty analysis of the measured global radiation was done and the internal estimate of uncertainty was evaluated following [32] and it was found that the value for uncertainty for the measured global radiation was 2.23%. The system included two modules in series, and the area of one solar cell is 0.0139 m^2. The number of solar cells in the two modules was 72. Therefore, the efficiency analysis of a PV system for its performance assessment is done here based on some experimental data as explained above (Fig. 5.14).

Using (5.47) through (5.52) and experimental data from Joshi, Dincer, and Reddy [33], energy and exergy efficiencies are calculated and shown in Fig. 5.13. It is clear from the figure that the energy efficiency (33–45%) is higher than that of the exergy efficiency (11–16%) of the PV/T system and (7.8–13.8%) of the PV system. Maximum exergy efficiency for the PV/T (16%) and PV (13.8) can also be seen at 4 p.m. where as a minimum exergy efficiency for the PV/T (11%) and PV (7.8%) is at 12 p.m. In the present study, natural air is used to derive the heat from the photovoltaic surface. However, if air is supplied beneath the photovoltaic surface by a forced mode (e.g., by putting a fan beneath the photovoltaic panel), more thermal energy can be removed in a better as well as convenient way. In that case a higher energy and exergy efficiency can be achieved.

Figure 5.15 shows the comparison of exergy efficiencies of both PV as well as PV/T systems. Comparing both curves one can see that the exergy efficiency of PV/T is on average 20% more than that of PV. Carbon dioxide, a major greenhouse gas, is responsible for global warming hence there is a need to understand the ways by which we can reduce such emissions. One solution to this problem can be adopting

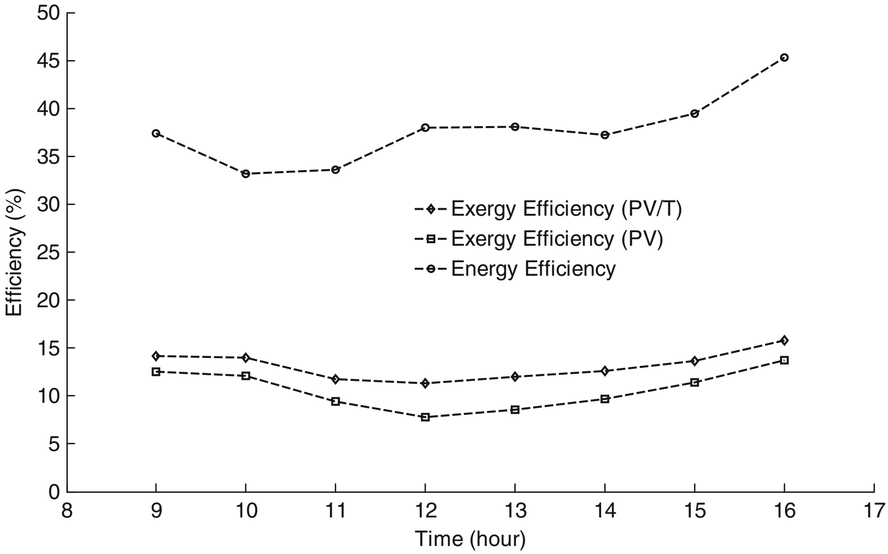

Fig. 5.14 Energy and exergy efficiencies of PV and PV/T systems (modified from Ref. [20])

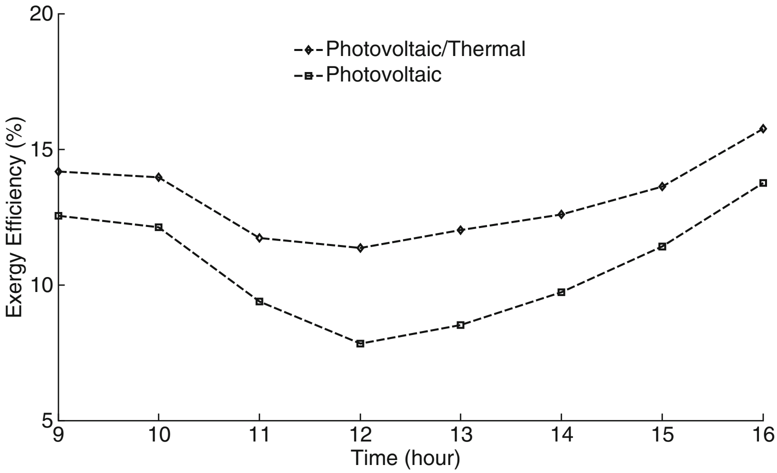

Fig. 5.15 Comparison of exergy efficiency of PV and PV/T systems (modified from Ref. [20])

unconventional energy sources wherever applicable; for example, for water heating one can use solar energy which is more eco-friendly as compared to using an electrical water heater that runs on electricity produced by conventional sources. Another example could be solar pumping: farmers irrigate their fields in the daytime and they can use solar water pumping instead of using an oil-based

generator to produce electricity and use it to run the water pump. The unconventional energy sources (often called renewable energy sources) are environmentally benign as they emit fewer greenhouse gases into the atmosphere as compared to conventional ones. Although the nonrenewable or conventional sources of energy such as coal, oil, and natural gas are more economical than the renewable sources, they pollute the environment at a much faster rate. Coal-based electricity generation causes the highest greenhouse gas emission among all the conventional and unconventional sources during the operation and the installation of a power plant.